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Supercritical CO₂ Brayton Cycles for Solar-Thermal Energy*

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Abstract

Of the mechanisms to improve efficiency for solar-thermal power plants, one of the most effective ways to improve overall efficiency is through power cycle improvements. As increases in operating temperature continue to be pursued, supercritical CO₂ Brayton cycles begin to look more attractive despite the development costs of this technology. Further, supercritical CO₂ Brayton has application in many areas of power generation beyond that for solar energy alone.

One challenge particular to solar-thermal power generation is the transient nature of the solar resource. This work illustrates the behavior of developmental Brayton turbomachinery in response to a fluctuating thermal input, much like the short-term transients experienced in solar environments. Thermal input to the cycle was cut by 50% and 100% for short durations while the system power and conditions were monitored. It has been shown that despite these fluctuations, the thermal mass in the system effectively enables the Brayton cycle to continue to run for short periods until the thermal input can recover. For systems where significant thermal energy storage is included in the plant design, these transients can be mitigated by storage; a comparison of short- and long-term storage approaches on system efficiency is provided. Also, included in this work is a data set for stable supercritical CO₂ Brayton cycle operation that is used to benchmark computer modeling. With a benchmarked model, specific improvements to the cycle are interrogated to identify the resulting impact on cycle efficiency and loss mechanisms. Status of key issues remaining to be addressed for adoption of supercritical CO₂ Brayton cycles in solar-thermal systems is provided in an effort to expose areas of necessary research.

Keywords: supercritical CO₂, Brayton, solar-thermal, concentrating solar power, energy

1. Introduction

It is recognized that solar-thermal energy can play a useful role in generating electrical power despite concerns regarding cost, as the thermal source is accessible and ubiquitous. One platform to produce power from a solar resource is using the point-focus, power-tower system in which the solar-thermal energy is concentrated thereby elevating the working temperature and associated efficiencies. Solar assisted power production to offset carbon emissions [1] and thermal storage for grid stability [2-4] remain strong motives for utilizing this approach. Cost-reduction efforts have been implemented to improve solar-thermal power production [5] with more aggressive efforts being supported by the U.S. Department of Energy [6]. High-efficiency power cycles is a critical component in achieving the cost reduction goals and may require temperatures that reach above 600 °C to obtain cycle efficiencies in the 50% range.

The supercritical carbon dioxide (sCO₂) Brayton cycle has emerged as a promising avenue for high-efficiency power production. With growing interest in renewable energy sources, cycles with high efficiency are critical to achieving cost-parity with non-renewable sources. Convergence on sCO₂ Brayton is occurring from the nuclear [7-9] and geothermal [10] fronts, in addition to solar-thermal [11-13]. Turbomachinery for sCO₂ Brayton is in the development phase [14-19] and is gaining momentum as interest grows and technical risks are reduced. However, adaptation of the cycle to interface with various heat sources will be imperative for its adoption as an industry-manufactured technology.

Over the past decade, there has been a significant amount of research on sCO₂ power cycles and heat transfer. Turbine and compressor performance characterization and prototype system testing has been a primary focus at Sandia National Laboratories (SNL) within the Advanced Nuclear Concepts group [17-23]. System control and transient analysis on sCO₂ has been a large focus at Argonne National Laboratory (ANL), specifically for Lead Fast Reactors (LFRs) and sodium-cooled reactors [24-29]. Various cycle configurations have been investigated for specific reactor designs [8, 9, 30-33]. Echogen has considered sCO₂ cycles for waste heat recovery, utilizing smaller power systems [34] and views sCO₂ as a valid competitor to steam technology [35]. While some work has been done pertaining specifically to solar applications, the literature for sCO₂ is introductory by comparison [11, 36]. The Southwest Research Institute is also active in enabling sCO₂ for solar energy and is pursuing turbo-expander and heat exchanger development for this purpose [37].

Interfacing the solar resource with a sCO₂ Brayton cycle requires a receiver to absorb the solar-thermal energy from the incident concentrated flux and transfer the energy to a transport media. The transport media in the receiver can either be the same as the power cycle working fluid (direct receiver) or employ a secondary media, either fluid or solid, that would experience heat exchange with the power cycle working fluid (indirect receiver). A direct receiver approach can leave the power cycle exposed to potential issues with a transient heat source whereas an indirect approach provides a buffer from transients.

This paper demonstrates the response of a prototype sCO₂ Brayton cycle under transient operating conditions similar to that experienced in a typical solar plant with a direct receiver. While the operating conditions of temperature and pressure for the experiments are lower than that desired for high-efficiency operation, this data serves to validate modeling efforts that can be used to evaluate higher-temperature systems. A discussion of primary mechanical and thermal losses is provided as well as areas of advancement required for adoption of sCO₂ Brayton turbomachinery for solar applications.

2. Experimental Loop

2.1 Layout

The experimental loop installed at Sandia National Laboratories (Figure 1, Table 1) is a split flow recompression cycle. The 'split flow' indicates that two separate turbines receive separate, dedicated flows. These two flow streams are expanded and then recombine after the turbines. A second flow split is located prior to the cooling and compression stages. One stream of the low-pressure flow is 'recompressed' without rejecting heat and is designed to operate at temperatures above the critical point. The main compressor operates near the critical point and the flow stream through this compressor experiences heat rejection. This configuration is expected to have improved cycle efficiency relative to a simple Brayton cycle. First, there is less heat rejection in the pre-cooler resulting in smaller heat loss as only a fraction of the flow passes through this component. Second, the thermal capacities of the hot and cold flows in the low temperature recuperator (LT recuperator) are better matched to optimize heat recuperation [38] with a resulting mass flow ratio of hot to cold that is close to 2:1. This is because the low temperature fluid exiting the main compressor is much closer to the critical point and therefore has a specific heat that is approximately double that of the higher temperature flow. Matching thermal capacities optimizes the heat transfer.

Figure 2a presents the thermodynamic state points for a representative recompression Brayton cycle capable at Sandia. The expansion process from points 5 to 6 is the same regardless of the number of turbines, assuming equal speeds for separate shafts. The final loop design with two separate turbo-alternator-compressors (TAC) was determined, in part, by a staged approach due to anticipated incremental government funding. The modular nature of the design allows for multiple configurations, a feature that enables proprietary and novel configurations by independent institutions.

The thermal input for the system is 780 kW and was selected based on key control and stability issues of the sCO₂ Brayton cycle while small enough to be affordable over several years of development. The main disadvantage of the relatively small size, and the resultant high turbomachinery rotational speeds, is that the system requires bearing, seals and motor alternator approaches that are not necessarily representative of a commercial-scale system. A compromise between fidelity and cost was achieved, while addressing the underlying questions for the technology to reduce risk for future industrial efforts in sCO₂ power systems.

2.1.1 Cycle Components

A complete description of the major components that constitute the Sandia split flow recompression test assembly is presented in Sandia report SAND2012-9546 [39]. The following is a summary description of the major components.

The TACs are hermetically sealed pressure vessels, rated for the maximum pressure and temperature conditions anticipated in the flow system. Within the vessel, the compressor wheel, gas bearings, and turbine are laid out along the shaft as shown in Figure 3. CO₂ enters the compressor on the right hand side of the compressor, and is discharged radially. Likewise, hot CO₂ enters down from the top of the turbine, and is expanded radially to the left-hand side of the figure. At design conditions, both turbomachinery wheels are subjected to pressures in excess of the critical point. During operation, leakage flow passes around the compressor and turbine through abradable labyrinth shaft seals to provide lubrication to thrust and journal bearings. The leakage flow is continuously pumped out of this region using scavenging pumps,

driving a cooling flow and maintaining reduced film pressures (ideally around 1.4 MPa) in the central cavity surrounding the permanent magnet shaft and bearings.

The split-flow Brayton cycle uses TACs to compress the low-pressure and low-temperature CO₂ to a high pressure at the compressors, and then expand the high-pressure and high-temperature CO₂ in the turbines. At and near design conditions, the turbines generate more power than the compressors and inefficiencies consume, and the remaining power is used to make electricity in the motor alternator. In the power generation mode, the alternator applies an electrical load to the TACs' rotating shaft that is sufficient to maintain the commanded rotational speed. The applied electrical load represents the power that the TAC would produce for consumer use. TAC-A takes, as input to the compressor, the flow that discharges from the gas chiller, which is the coldest point in the circuit. As such, it is also the least compressible. Therefore, TAC-A consumes less energy per unit mass to compress the fluid than the recompressor in TAC-B.

A single low-pressure flow discharges from the LT recuperator, where it splits into two flow paths, one path to each compressor. The fraction of the total flow going to each compressor is a function of the relative speeds of the two TACs and the thermodynamic state of the fluid at each inlet. These factors combine to determine each compressor's discharge pressure. When the two flows recombine (at the main compressor flow discharge from the LT recuperator) they must be at the same pressure. Pressure mismatch at this point can put a compressor into a potentially damaging state of surge. The primary control to avoid surge is the speed of each TAC, with the magnitude of heat rejection in the gas chiller being of secondary importance. Minimum and maximum rotational speed for each TAC is 25,000 rpm and 75,000 rpm, respectively.

A recompression cycle requires two recuperative heat exchangers, referred to here as high temperature (HT) and low temperature (LT) recuperators. The total low-pressure flow exiting the turbine flows through both recuperators, while the total high-pressure flow exiting the compressors flows through the HT recuperator (only the flow exiting the main compressor passes through the LT recuperator). A third heat exchanger is required to reject heat to maintain the system operating point. Heatric printed circuit heat exchangers (PCHE) were selected for all heat exchanger components. The flow passages in PCHE's are etched into 316 stainless steel plates, which are then stacked and diffusion bonded to form the core of the heat exchanger. The resulting components enable an extremely efficient and compact heat exchange. The HT and LT recuperators are designed to transfer 2.3 MW and 1.7 MW, respectively. In the recompression configuration, the LT recuperator duty is only 0.6 MW. The heat rejection PCHE is rated at approximately 0.54 MW, which is sufficient to establish the main compressor inlet CO₂ temperature near the critical temperature.

The solar heat source is simulated using electrical resistive heating in six shells through which the CO₂ flows. Each of the six immersion heaters provides 130 kW of heat input, providing a total heating capacity of 780 kW. With these heaters, there is sufficient power to reach temperatures up to approximately 538 °C.

2.2 Operation

The primary objective of testing in this work is to achieve steady operating conditions and then observe the system response to perturbations in operating conditions, thereby simulating a transient solar environment. When operating a power cycle in combination with a solar resource, the potential exists that the direct normal insolation (DNI) used to heat the working fluid may decrease drastically during periods of cloud cover. These transients may last for as short as seconds or as long as hours and days. While thermal storage can mitigate the effects of

drastic transients, usage of thermal storage to level the power output will be dependent on market conditions that drive power plant operation, availability of storage and the short-term forecasting of the cloud transient. Due to diurnal cycling, even without cloud transients, power cycles driven by a solar resource are inherently transient. Stability may be improved through usage of other power sources such as nuclear, geothermal, etc. However, characterization of transient responses is still required for off-normal operation especially as one considers the fluctuations around the CO₂ critical point.

Test runs of the recompression Brayton cycle are time-intensive. Test preparation includes evacuating the system over night to near-zero pressure followed by filling with CO₂ to the desired mass loading, and then elevating the system temperature from a cold state to the selected steady state operation. The current maximum thermal ramp-up rate is approximately 5 °C per minute. At this rate, increasing the operating temperature from 17 °C to 477 °C requires approximately 2 hours. This ramp rate is derived from experience and by piping stress limitations; alternate heating rates and associated designs can be optimized for particular system needs.

In order to simulate the system response to a fluctuating heat source, several test runs were conducted where the heater power settings were reduced by 50% and 100%. The time periods for 50% reduced power setting (Figure 4) are from 7012 to 7073 seconds, and from 7349 to 7529 seconds (61 seconds and 170 seconds, respectively). The time periods for 100% reduced power setting (Figure 5) are from 4082 to 4148 seconds, and from 5332 to 5465 seconds (66 seconds and 134 seconds, respectively). Before adjustment of the power setting, the nominal heating power inputs for the 50% and 100% cases are 280 kW and 160 kW, respectively. It is difficult to maintain a perfectly steady state condition prior to transient excursions due to variations in TAC speeds and system cooling, among other perturbations. A best effort was made to establish conditions prior to an excursion that would produce accurate indications of a true system response to simulated solar resource transients.

System response is characterized in four separate plots for each heater power setting reduction, namely: pressure response in the low-pressure and high-pressure legs, temperatures at the heater inlet and discharge, and the system net power generation response. Negative power indicates power production by the system. The large change in net power generation indicated at 4870 s in Figure 5d are due to a controlled reduction in cooling water temperature, which caused the compressor inlet flow to become more dense, less compressible, and therefore requiring less compressor work. The main compressor experienced an increase in mass flow, at the expense of the recompressor, but at a higher density. The greater mass flow at lower compressibility resulted in a roughly net-zero change in power for the main compressor. However, the reduced mass flow to the recompressor resulted in a sharp decline in required compressor power.

The mass associated with the heat input system results in a thermal capacitance effect. Despite changing the heater power setting by a 50% or 100% reduction in power, the thermal input to the cycle fluid does not necessarily reduce by 50% or 100%. Instead, for the 50% reduction, the thermal input (due in part to thermal capacitance of the piping and heater array) declined to minimum values of 210 kW in the first excursion, and 200 kW in the second. This thermal input was determined by using the enthalpy change in the fluid across the heaters and the mass flow rate. For the 100% power reduction, the thermal input (all due to thermal capacitance) declined to minimum values of 60 kW and 56 kW in the first and second excursions, respectively.

From plots (a) and (b) in Figure 4 and 5, it is apparent that system pressures decline in response to the loss of thermal input. An inherent characteristic of a closed Brayton system is that as the heated cycle fluid increases in temperature and decreases in density during startup, it pushes fluid to the colder components, effectively increasing the whole system pressure. Thus, when the hot side declines in temperature, so too will the system pressure.

The low-pressure leg response to thermal power input reduction is a modest decline, approximately 50 kPa or less. The high-pressure leg response (plot b, Figure 4 and 5) is greater, with a maximum reduction approaching 100 kPa. Thus, the high-pressure leg responds to thermal input changes with greater fidelity.

The cycle fluid temperature response is presented in plot (c) of Figure 4 and 5. These plots for the 50% and 100% reductions show the temperatures immediately downstream of the power perturbation (heater discharge), and immediately upstream (heater inlet). As one would expect, the temperature immediately downstream shows a much more dramatic and immediate change than the upstream temperature. Downstream and upstream temperature reductions for the 50% power reductions are 20 °C and 10 °C for the first transient, and 35 °C and 15 °C for the second transient. The corresponding upstream and downstream temperature reductions for the 100% power reductions are 40 °C and 10 °C for the first transient, and 50 °C and 15 °C for the second transient. The upstream response is muted and delayed for several reasons. First, there is a finite period of time required for the fluid to transit the loop. The fluid that is immediately affected in the heater requires that transit period to return to the heater inlet. Second, the recuperators inherent in the design of a recompression system act to mitigate the sharpness and magnitude of thermal changes in the cycle. Finally, the thermal capacity of all piping and components acts to delay the magnitude of a thermal transient.

In general, the various responses in the pressure, temperature and power output to the reduction in thermal power input exhibit an exponential decay, indicative of what is expected from stored thermal energy in a thermal capacitor. System response after the heating power is restored exhibits a complementary logarithmic rise. These trends are best displayed in the temperature histories in plot (c). When solar transients do occur, short perturbations of this type can easily be managed by thermal capacitance in the system with the extent of the exponential decay in system variables dependent on the total thermal mass and heat losses inherent in the heating system.

2.3 Measurement Uncertainty

A particular test case and operating conditions was selected for consideration of system losses and measurement uncertainty. A data point 7600 s into the test was selected from Figure 4 and analyzed at a time of steady power generation (see also Table 2).

2.3.1 Thermal Loss

Thermal losses between two separate locations in the cycle can be assessed by examination of temperature change across lengths of piping that are not directly heated or cooled as part of a cycle process. This is most significant along the high-temperature legs of the system. These segments are larger in diameter and are of considerable length to accommodate thermal expansion. The loop structure continues to be developed and is largely not insulated. This is responsible for a significant amount of the poor system performance from the standpoint of total system efficiency, providing context to discrepancies between theoretical and experimentally observed cycle characteristics.

Figure 1 indicates the locations of significant heat loss. Heat losses are estimated in these regions by a product of mass flow, heat capacity, and temperature change. For the selected test condition at 7600 s (Table 2), 62.3 kW of heat is lost from the HT recuperator to the heater inlet, 40.9 kW is lost upstream of the turbines, and 10.5 kW is lost from the turbine outlet legs to the HT recuperator inlet. Losses at the turbine volute are also noted to be 8.8 kW and 7.5 kW for turbines A and B, respectively. Volute thermal losses are estimated from known conditions at the inlet and outlet of the turbine, and turbine performance maps for a given set of conditions [40]. The discrepancy between measured turbine outlet conditions, and outlet conditions predicted by the performance map is attributed to a cooling mechanism at the back of the turbine volute due to rapid expansion of high pressure CO₂ across the rotating seal, into the low-pressure alternator housing. In total for this test case, 130 kW is lost to various thermal mechanisms for a heater input of 342 kW (38%).

2.3.2 Losses to Rotating Friction

Rotating loss, or windage, is also a significant contributor to conversion inefficiencies for the current test assembly. The high-speed environment, along with high density and low viscosity, generate a highly turbulent environment at the shaft and within tight clearances of the gas foil thrust and journal bearings. The presence of turbulence causes a sharp increase in the dependency of frictional loss and load capacity to environmental conditions, namely a heightened sensitivity to lubricant gas pressure and runner speed. This phenomenon was first observed in testing of journal bearings at NASA's Glenn Research Center [41]. Intensive frictional losses not only serve as a parasitic load, but also can cause extreme local heating of bearings and other turbomachinery internals, causing turbine malfunction.

Modeling TAC windage has revealed that the major loss is due to the shaft itself and the thrust bearing assembly; these combine for upwards of 85% of frictional loss [38]. The two radial bearings account for the remainder. These estimates are based on consideration of the turbomachinery assembly as a simplified series of tightly-housed concentric cylinders and disks and applying friction relations for turbulent boundary layers by Schlichting [43] and Vrancik [44]. Figure 6, illustrates the relative contribution of windage loss (based on these turbulent correlations) for the journal bearings, thrust bearing and rotating shaft. The sum of these power losses is also provided in the figure. Ongoing work seeks to optimize the thrust bearing assembly for reduced friction and resistance at higher temperatures without compromising load capacity. This is the primary challenge in attaining high speeds approaching 75 krpm for the current test assembly.

To isolate and quantify empirical rotating losses for the present test assembly, a series of tests was conducted to identify the net contribution of the windage losses (illustrated in Figure 6) plus that due to seals. This total rotating loss was measured by removing the turbine and compressor wheels from the shaft, and recording the resultant power consumption required to spin the bare shaft alone at high speeds within prototypic CO₂ environments. Sensitivity to CO₂ properties, shaft speed, and thrust loads were evaluated directly [42]. CFD modeling of the sCO₂ lubrication layer was also undertaken to confirm that observed losses were consistent with turbulent theory. The resulting correlation for power loss that scales with angular velocity (ω) and fluid properties (density ρ and viscosity μ) is as follows:

$$P_{turb} = 0.155\omega^{2.8} \left(\frac{\rho}{21.1[kg/m^3]} \right)^{0.8} \left(\frac{\mu}{14.9[\mu Pa \cdot s]} \right)^{0.2} \quad (\text{Eq. 1})$$

Equation 1 can be used for estimating windage losses in test data within +/-5% for speeds less than 50 krpm. This correlation runs approximately 20% higher than that predicted by turbulent theory (Figure 6) for rotating disks and cylinders alone [42]. This can be attributed to the simplified modeling approach and neglect of shaft seals. Empirical testing results yielding equation 1 indicates an estimated 4.9 kW loss for turbine A, and 10.1 kW loss for turbine B. The difference between A and B in this case is due to their speed differential at 7600 s and fluid properties in the rotor housing. Here, CO₂ properties of density and viscosity are taken within the turbine housing, where the shaft and gas bearings operate. Typical temperatures and pressures in the rotor housing are 150 °C and 1.4 MPa.

2.3.3 Leakage Flows

At each turbine and compressor wheel, leakage flow bypasses the rotating element in the housing through abradable labyrinth shaft seals and into the turbine housing where the gas bearings and alternator spin in a reduced pressure environment (Figure 3). The leakage flow critically provides bearings with a hydrodynamic film for load support and transfers frictional heating. A gas scavenging system pulls CO₂ from the housing to prevent buildup of pressure, driving a cooling flow through the turbine, and pumps it back into the high pressure loop to complete the closed cycle. A supplementary bypass line connects from the compressor inlet at each TAC unit and penetrates into the turbine housing in the vicinity of the high temperature turbine-end radial bearing. This additional cooling is metered by a manual needle valve along the flow path.

The combination of leakage flow in the seals and bypass flow for cooling bearings is considered the total system leakage flow. The mass flow, temperature and pressure of this flow is measured and considered in the data reduction process as follows. During the compression stage, leakage flow in the seals is included in the compressor work calculation, since it is judged that the flow cannot reach the seals without first passing through the centrifugal wheel. During expansion, leakage flow is not included in the turbine work calculation, since the leakage flow largely bypasses the turbine wheel.

The current approach for seals limits leakage flow to less than 5%. Consequently, the impact to compressor and turbine work calculations is minimal, but can still represent up to 5% of unrecoverable loss. Given the nature of abradable seals and the small-scale geometry of the present rotating hardware, this flow rate may change somewhat over time for a fixed set of CO₂ conditions upstream of the seal, as the labyrinth seal experiences wear. For this test on 9/11/2012 near 7600 seconds, the total mass flow topped 3.5 kg/s while leakage flow was 0.1 kg/s in total, or 2.8%.

2.3.4 Uncertainty Analysis

Analysis of net power generation requires knowledge of compressor and turbine work (as measured by enthalpy change across each active component), and estimated thermal and frictional losses. Each has associated uncertainty. For the compressor and turbine evaluations, calculations depend on measurements of mass flow and enthalpy change across the component of interest. Using an in-house data reduction and analysis code, enthalpy is evaluated by using the RefProp [45] property tables based on local temperature and pressure at the inlet and outlet of each component. An exception to this is that measured density is used for the enthalpy calculation at the compressor inlet, which is near enough to the saturation region that temperature and pressure alone cannot be relied upon.

Uncertainty (U) in compressor work, for example, can be calculated based on partial derivative of work (W) with respect to each input into the calculation (x_i), using the following relative instrument uncertainties: temperature +/-1%, mass flow +/-4%, pressure +/-5%, and density +/-1%.

Propagation of uncertainty was carried out by evaluating each partial derivative and combining them as shown in Equation 2.

$$U_W = \sqrt{\sum_{i=1}^n \left[\left(\frac{\partial W}{\partial x_i} \right)^2 U_{x_i}^2 \right]} \quad (\text{Eq. 2})$$

For the particular test in Figure 4 (at 7600 s), each turbine work uncertainty was calculated with measured uncertainties to be 2%. The main compressor work, which operates nearest to the critical point, also exhibited an uncertainty of 2%, while the recompressor work is accurate to within 4%.

3. Cycle Modeling

A Fortran model of the Sandia split flow recompression Brayton cycle has been developed to investigate performance trade-offs and inform improvement decisions. The inputs to this model include main compressor inlet pressure and temperature, speeds for both TACs, and heater discharge temperature. Digital versions of the turbomachinery performance curves are interrogated, which requires an iterative approach to resolving a balanced steady state condition. Pressure losses throughout the loop are based on curve fits of experimental component pressure losses as a function of mass flow. Using these five inputs, a balanced operating point is obtained that defines the state points around the system, and component and system performance.

An assessment of the model fidelity relative to measured data was calculated. A significant challenge in applying this model to current test data is that the main compressor in TAC-A has been replaced with a recompressor wheel, which is designed to operate at conditions that the TAC-B recompressor wheel experiences. The primary difference is the inlet temperature. The recompressor wheel is designed for inlet temperatures approximately 28 °C higher than that entering the TAC-A compressor. The result of operating the recompressor at much lower temperatures than design is significantly reduced accuracy when interrogating the recompressor performance maps. Thus, predictions from the model in the vicinity of the TAC-A recompressor deviate from test data. Stated succinctly, fairly significant deviations between data obtained to date and model predictions are to be expected. Additional efforts to improve this prediction are underway and will come largely from expanding the envelope of speed, temperature, pressure, and power production experience.

In Table 2, the ‘Measured’ column presents the state points and resulting cycle performance at 7600 s into the test run from Figure 4 on 9/11/2012 and serves to compare actual system performance with predictions from the model of the test assembly including turbomachinery performance calculations based on boundary conditions. Table 2, column ‘Calculated (a)’ presents model predictions using actual test data measurements as inputs for only the five input parameters above. Comparison of the state points throughout the loop as well as the cycle performance parameters show good agreement and are adequate to have confidence in

extrapolating to different operating conditions. In particular, the predicted cycle efficiency is 6.2% at a measured efficiency of 5.3%.

Table 2, column 'Calculated (b)' is the model prediction output for the original design conditions of Sandia's recompression Brayton cycle. In this model run, the main compressor wheel has been installed and model input values have been set to the design conditions. Therefore, these predictions are expected to reliably predict the performance of the current loop. The original expectation for design performance included approximately 250 kW of electricity at an efficiency of about 32%. However, the model predicts 135 kW and a cycle efficiency of 15.2%. The deviation between original design performance and the predictions listed under the column labeled 'design' are directly attributed to heat loss, leakage and windage that were excluded from original design predictions. In addition, actual pressure losses throughout the system have been found to be greater than the original design pressure losses.

Thermal losses occur in the turbine housing, driven by the temperature difference between the hot turbine inlet volute and the water-cooled alternator compartment immediately adjacent to the turbine volute. Temperatures in the alternator volume are typically on the order of 100 °C or less. This is dramatically lower than the turbine inlet temperature, with a separation of only a few centimeters. This situation causes a large temperature gradient that drives thermal conduction losses from the turbine volute. The fluid temperature at the radial turbine wheel inlet is not currently measured. However, an attempt is made to quantify this loss by reducing the temperature used to interrogate the turbine performance curves until the predicted discharge temperature is sufficiently close to the measured discharge temperature. It is for this reason that the temperatures for points 5b-A and 5b-B in the 'Measured' column are about 4 °C less than the adjacent 'Calculated (a)' column in Table 2.

Columns 5 and 6 in Table 2 present model predictions for the same design system, but with accumulating improvements to the system. These include installation of insulation to eliminate thermal losses ('Calculated (c)'), and an improvement in compressor design to obtain efficiencies in the mid 80% range ('Calculated (d)'). With both of these improvements simulated, the model predicts power production of 172.6 kW, and a gross efficiency of 24.1%. Continued improvements can be predicted assuming reduction of other losses, such as leakage flows, windage, and pressure losses.

Other potential improvements to the cycle including raising the design temperature and pressure are possible. Currently, the installed system is limited by the maximum design temperature and pressure ratio of 538 °C and 1.8, respectively. A Brayton cycle designed to operate with a solar energy source can operate at temperatures in excess of 600 °C with pressure ratios on the order of 2.5. These changes greatly increase the thermal to electric conversion efficiency, and will likely yield efficiencies in the vicinity of 50% as is commonly cited [46].

4. Advances for sCO₂ Brayton adoption

There are a number of technical challenges that require attention for sCO₂ Brayton adoption in solar-thermal power generation. They include a significant amount of work in development of turbines, bearings, seals, heat exchanger design (especially when considering salt to sCO₂ heat exchange) and materials. The following outlines several of the needs associated with this technology adoption and are addressed here in an effort to provide a realistic view regarding the path to implementation as well as a projection of plant cost.

4.1 Turbine development

Several improvements in the Brayton turbomachinery could significantly improve the efficiency by reducing losses. A 10 MW_e system size has been identified as the minimum needed for demonstration of commercial-scale turbomachinery technology for sCO₂ [47]. Whereas the present test assembly uses radial compressors and turbines, a >10 MW_e sCO₂ power system would be a multi-stage axial machine much like present-day industrial gas turbines. The 10 MW_e size would also allow for high efficiency at rotational speeds on the order of 24,000 rpm, in comparison to the current 75,000 rpm system. This reduction in design speed to 24,000 rpm would allow the turbine to be mated with gear reduction to 3,600 rpm (60 Hz) using a commercially-available gearbox and allow for synchronous operation similar to commercial-scale power systems.

A significant drawback of the current design is that a simultaneous high-speed and high-pressure environment at the rotor can cause disproportionately large frictional losses. Also for this small-scale hardware, mere inches separate the high temperature turbine inlet (potentially up to 540 °C) from the motor/generator cooling water flow at room temperature, resulting in thermal losses. Leakage flow bypasses the turbomachinery wheels to lubricate the gas bearings, reducing productive work as well. To manage leakage flows, the turbine itself and other turbomachinery internals must be designed and built to tolerances on the order of thousandths of an inch. All loss mechanisms would be eliminated or greatly reduced for a commercial-scale (multi-MW_e) Brayton system [47].

Additional objectives include demonstration of high conversion efficiency and control concepts of the sCO₂ closed Brayton cycle. The current split flow recompression turbomachinery will be used to investigate system control algorithms for a recompression cycle. Variables of particular interest include responding to heat input variations, changes in load demand, and responding to emergency events. Primary control features will include turbomachinery speed, heat rejection, and possibly circuit mass loading. Developing control algorithms is necessary to maintain a recompression cycle at optimum performance with automatic controls to respond to various transients such as load demand and heat input. The current research and development system requires continual oversight to maintain the system in a stable condition.

4.2 Heat exchangers

When targeting high-capacity factors, a secondary thermal transport media for use in the receiver and storage system (indirect system) is a likely approach given the high pressures associated with supercritical working fluids and low specific heat values for gases. In this indirect system, the primary fluid is defined as that employed by the power cycle and the secondary fluid as that used for collection and/or storage. An indirect approach with different receiver and power cycle fluids allows the media used in each subsystem to be optimized for their specific function in the cycle components. This approach has an additional benefit of being applicable to a variety of power cycles, as the collection and power cycle media are decoupled. However, this requires a heat exchanger at the interface of these subsystems.

An important consideration relative to indirect systems is the ability to incorporate significant energy storage. In instances where capacity factors are relatively low (25%–50%), a direct receiver approach, which incorporates heat exchange to a storage media, can be beneficial because the majority of annual energy generation occurs with the heat-transfer media being sent directly to the power block without the incurred losses in a heat exchanger. For larger-capacity factors, where a significant fraction of the collected energy is sent to storage, it is more

efficient to match the storage and receiver media, thus, requiring only a single heat exchanger to interface with a separate power block working fluid (see Figure 7). For the calculations in Figure 7 based on a turbine inlet temperature of 700 °C and a 98% storage efficiency, the first-law efficiency for an indirect liquid receiver is higher than the direct approach, above approximately ten hours. The crossover in second-law efficiency occurs at a lower storage capacity of approximately seven hours. While this amount of storage capacity may appear to be higher than desired in some energy markets, the cost and feasibility of storing supercritical fluids that reach goals applicable to SunShot would prove cost-prohibitive for both large- and small-capacity factors [48]. Further, the effects of thick-walled piping and a potential slower start-up due to a larger thermal capacitance in system components has been neglected for the direct CO₂ approach, further justifying an indirect approach, especially for large capacity factors.

In order to successfully implement an indirect system for solar, a heat exchanger to transfer heat between the dissimilar fluids is necessary. Of the heat exchanger designs to consider, a diffusion bonded heat exchanger (such as a PCHE or hybrid-PCHE [49, 50]) is a possible candidate as the diffusion bonding process is capable of producing small channel sizes that enables containment of the high pressures required for the supercritical phase [51, 52]. A reduction in the channel hydraulic diameter also enables an increase in the heat transfer coefficient, as they are inversely related ($h \sim 1/D_h$). Thus, the small channel size accommodates the two major requirements for heat exchange with sCO₂.

On the salt side of the heat exchange, the same channel dimensions used for sCO₂ are not optimal due to the concern for plugging of solidified salt. Therefore a hybrid construction using techniques other than printed circuit methods may be required for salt applications [53]. When selecting salts that have higher operating temperatures, typically this also involves a corresponding increase in the melt temperature, making solidification problematic at temperatures well above ambient. Similar concerns exist for sodium in PCHEs, with initial investigations recently appearing [33, 54, 55]. One mitigation strategy is simply to utilize the salt only in a thermal environment where the temperatures never reach solidification temperatures. This is possible for recuperated Brayton cycles where the heat addition from an external source is expected to raise the temperature from 531 to 700°C [30]. Common chloride-based salts (e.g., KCl-LiCl-NaCl ternary eutectic melts at 346 °C [56]) or carbonates typically have melting temperatures well below this range. Start-up procedures, however, may need to involve external thermal input for preheating before salt introduction.

The sCO₂ Brayton cycle is known to be highly recuperative, with projected capital costs of heat exchangers representing 80% of the total cost of the cycle [31, 57]. Highly compact, efficient heat exchangers are, therefore, necessary for power block cost reduction with numerous design and characterization studies in the literature [58-71]. However, there is very little information on heat exchanger design when considering exchange with a secondary hot working fluid, such as liquid metals or molten salts [54, 68]. When considering liquid sodium on the hot side, initial studies have only begun to understand conditions under which freezing conditions may occur [54]. The implementation of a salt-to-sCO₂ heat transfer interface must address the differential pressure between the hot and cold fluid which may be as high as 20-25 MPa at temperatures of 600-800 °C; not a trivial matter. Idaho National Laboratory (INL) has initiated work relative to using salt-service heat exchangers to link a sCO₂ Brayton cycle to a nuclear reactor. Preliminary studies for fluoride salts indicate that shell-and-tube (helical coil) and PCHE heat exchangers are the most likely to achieve the desired results for their advanced high-temperature reactors [51, 52], with the PCHE option preferred for its thermal and structural performance. Further, they

demonstrated diffusion bonding for nickel-based alloys Haynes N and 242, despite the high chromium content in these materials resulting in protective oxide layers [49]. Argonne National Laboratory has initiated sodium-to-sCO₂ heat exchanger studies with plans to test fill and drain operations to avoid channel blockage [53].

In solar plants where molten salt is the industry standard, it has been long known that molten salts (nitrates and halides) tend to preferentially dissolve and deposit active alloying elements, (e.g. chromium or molybdenum [72-76]), which would need to be addressed for smaller channels where plugging could be an issue [75]. By forcing the salt toward a reducing condition, corrosion becomes extremely slow and may sufficiently inhibit this behavior [77]. sCO₂ corrosion of metals is currently being investigated by several institutions with stable oxides (chrome and nickel oxides, and alumina) as protective barriers [78]. It has been found that high concentrations of chromium and nickel significantly increase the corrosion resistance of steel alloys in CO₂ [79]. Current studies include investigations of protective layers exposed to impurities and developing protective barriers [80].

4.3 Bearings/seals

To date, the approach to gas bearings and seals for this system has demanded a disproportionate amount of the total research effort. The closed cycle, small-scale turbo-alternator-compressors developed for the present demonstration loop are a result of custom fabrication, and an iterative design and testing process. This has resulted in a system capable of supporting the necessary thrust loads, on the order of 400 N, at high speeds, within minimal irreversible losses. However, there is still considerable room for improvement. Modeling results indicate that a bearing with smaller diameter and fewer thrust pads could maintain thrust load capacity with reduced frictional losses. In addition, incorporation of geometric features (e.g. chevrons) at the trailing edge of each thrust pad to expel hot fluid would likely increase load capacity by enhancing the thermal wedge effect, while improving thermal management [81]. Experimental work has also demonstrated that using a stamped manufacturing approach rather than assembling the thrust bearing manually from many small pieces can attain tighter engineering tolerances. Smaller engineering tolerances allow for operation at reduced film thicknesses, increasing load capacity.

Finally, it is recommended that shaft and thrust runner of next generation CO₂ gas bearings be plasma sprayed with a solid lubricant while using bare pads. This improvement is directed at increasing the temperature resistance of the current model, which cannot be heated beyond the dissociation temperature of Teflon.

Commercial-scale systems would apply a different approach to bearings and seals. These systems would almost certainly be large enough to operate efficiently at 3600 rpm (60Hz), eliminating the feasibility of high-speed gas bearings. A commercial generator would be located outside of the high pressure CO₂ region, likely by using dry liftoff seals to separate the rotor from ambient conditions. Industrial dry liftoff seals use several stages and a buffer or purge gas to isolate the working fluid from the environment, resulting in reduced friction from the present assembly, and use of standard oil-lubricated industrial bearings types.

4.4 Materials

Material requirements for thermal solar power applications vary widely depending on the heat transfer fluids under consideration and operating conditions imposed. Focus here will briefly discuss the material requirements of CO₂, oxoanion salts (nitrate/nitrite and carbonate), and

halide anion salts (fluorides and chlorides) that could be used as primary or secondary heat transfer fluids depending on the receiver, power cycle and thermal storage subsystem configurations. Nitrate/nitrite salts are currently used in commercial solar applications, but there is concern with the thermal stability above 600°C, thus other fluids must be considered for higher temperatures. Carbonates and halides have also been selected for consideration based on their high temperature stability and cost.

Materials with the ability to form passivated oxide layers, such as a chrome oxide or alumina, have been found to perform well with CO₂ [78, 82]. Quantification of the presence of impurities (e.g. moisture) and their role in exacerbating corrosion is necessary for long-term power plant operation [82]. While it is understood that an aggressive attack on containment materials will occur in the presence of impurities there are no well-defined limits that currently exist.

Oxoanion salts, specifically molten nitrate/nitrites and carbonates, have different material considerations than that of CO₂. It has been observed that nitrate/nitrite salts and carbonate salts are able to form and maintain passive oxide barriers that are thermodynamically stable in the melt, which act as diffusion barriers that form following typical parabolic growth rates [83, 84]. In contrast to CO₂, active alloying elements, such as chromium, are soluble in the melt. Corrosion enhancing impurities typically take the form of chlorides, which act to disrupt passive layers and act as a catalyst for corrosion and must be considered from a systems engineering standpoint [85-87]. Thermal decomposition of oxoanion salts into oxides will increase the basicity which, in turn, changes the thermodynamic state of the melt. This decomposition is reflected in potential-oxide (E-pO₂-) diagrams (which parallel Pourbaix diagrams for aqueous solutions) and indicates potentially stable phases within the melt, useful in predicting phases that may be used as a protective oxide layer [88]. Questions related to evolved oxide content over time (i.e. thermal decomposition of a given salt) for the long-term stability of the salt need to be addressed, in addition to techniques of online monitoring of salt chemistry.

Halide salts differ significantly from oxoanions in that they do not form passive oxide layers, as is the case with chlorides [89] and fluorides [90-93]. In the case of fluorides, a metal fluoride is more stable than the metal oxides. Alloy protection with fluorides must rely on thermodynamic equilibrium between alloys [94] and this approach has largely been used with chloride melts. Due to the lack of a diffusion barrier, corrosion-enhancing impurities in halides take the form of oxygen or oxygen containing molecules, such as water or air [95, 96]. Systems' where initial salt purity and ullage gasses are not controlled experience severe corrosion [77, 97]. Systems' using these salts requires monitoring and purification systems in order to control corrosion of containment vessels. Questions are still outstanding related to chloride systems as to the practical development of thermodynamically and kinetically favorable oxide barriers that might lessen requirements of salt purity, which may preclude the need for a pressure vessel in potential system designs. Information on corrosion rates are incomplete and poorly controlled in many studies, this lack of information on the kinetics of corrosion will be required from a systems standpoint.

Diurnal cycling within a CSP plant places an increased emphasis on materials resistance to cycle fatigue failure. Studies on heat exchangers, for nuclear applications, have focused on the overall strength making alloy 617 a logical choice [98, 99]. The introduction of thermomechanical stress in a CSP facility motivates evaluation of low cycle fatigue (LCF) properties. Haynes 230, a nickel alloy with high tungsten content, has excellent fatigue life characteristics. As a comparison, Haynes 230 has been observed to fail around 50,000 cycles at 760°C, while 617 fails around

15,000 cycles [100]. This is a dramatic difference and will be important for selection in high temperature receiver materials at a minimum for the next generation of solar power plants.

4.5 Impact on cost goals

Comprehensive, critically reviewed costing data for the major components of a commercial sized (~ 10 MWe) recompression, closed-Brayton cycle for CSP applications were not found in the open literature. Private parties interested in developing components consider costing data proprietary. To fill this cost information void, Sandia National Laboratories, Oak Ridge National Laboratory (ORNL), and the Department of Energy (Office of Nuclear Energy) are initiating research into these various costs. The result of this effort is intended to be a modeling tool that predicts the levelized cost of electricity (LCOE) for these systems and incorporate the information into the current LCOE program that ORNL maintains. As the working fluid operates in the supercritical phase, the turbomachinery size [35] and cost may be lower but extensive recuperation will necessitate costly heat exchange.

Assuming similar power block costs as that for steam-Rankine, the estimated impact of the proposed Brayton cycle to achieve SunShot goals is assessed using the National Renewable Energy Laboratory's System Advisor Model (SAM, <https://sam.nrel.gov/>). Assuming a molten salt power-tower plant model and adjusting the power block efficiency (54% gross) and receiver temperature (700 °C salt) to account for an indirect, dry-cooled, sCO₂ Brayton cycle, the LCOE for a 100 MW_e system has been estimated. In order to achieve 6 ¢/kWh (real), a solar multiple of 3.1 and 16 hours of storage is required for the following SunShot-driven system:

- Cost assumptions: site preparation = 10 [\$ /m²], solar field = 75 [\$ /m²], power plant = 1,160 [\$ /kW], tower/receiver = 150 [\$ /kW_t], thermal storage = 15 [\$ /kW_t], contingency = 0%, indirect (sales tax and land) = 17.8%, interest during construction = 6.0%, O&M = 40 [\$ /kW_{yr}].
- SunShot financial assumptions: discount rate = 5.5%, inflation rate = 3%, debt rate = 6%, state income tax = 5%, return on equity = 15%, debt fraction = 62%, federal tax = 35%, depreciation = 5 yr MACRS, ITC = 0%.

Of particular note are the large solar multiple and 16 hours of storage required to achieve the desired SunShot LCOE target. This results in a capacity factor of 73.1% and is in the range that a direct CO₂ receiver/system cannot efficiently provide (see Figure 7). Capacity factors of this magnitude must use the storage media in the receiver, thereby requiring liquid-to-sCO₂ heat exchangers of the type pursued in this proposal for implementing sCO₂ Brayton to capitalize on the cycle efficiency benefits.

5. Conclusions

The sCO₂ Brayton cycle has been shown to have significant efficiency benefits especially as solar-thermal power plants increase their operating temperatures above 600 °C. In particular, part heat load operation, common to a solar resource, appears manageable especially for short durations (e.g. short-term cloud cover) due to thermal capacitance in the system and piping. Therefore it is recommended that sCO₂ Brayton cycles continue to be pursued for solar-thermal energy applications.

Thermal, rotational and mass losses for this prototype cycle have been quantified and utilized to benchmark a cycle model with good agreement. Measurement of the system performance

indicates an efficiency of approximately 5% for the operation conditions selected in this work. At design conditions, this efficiency is expected to increase to 15% using the benchmarked model presented in this work and would approach approximately 24% with minor modifications to improve insulation.

Predicted efficiencies still far short of the 50% thermal efficiencies claimed for sCO_2 at a 600 °C turbine inlet temperature. This is primarily a limitation of the laboratory-scale demonstration turbine used for the present study. At around 250 kW_e in size, the test facility was intended to be large enough to confront the fundamental issues for sCO_2 Brayton cycle technology, but small enough to be affordable over several years of incremental funding. For adoption of sCO_2 in a solar-thermal power plant, a number of required advances remain and are largely addressed by moving to larger equipment in 10 MW_e range. For large capacity factors and indirect systems, heat exchange between CO_2 and a secondary fluid amenable to solar is also required. This represents a significant challenge in terms of material selection as well as heat exchanger design. Improvements in bearings and seals to prevent leakage are also required, especially as the system scales up from the small prototype included here.

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Table 1. Nominal performance values corresponding to Brayton cycle components shown in Figure 1.

Parameter	Design Point Values
Heater input	780 [kW]
Pre-cooler	531 [kW]
Turbine A efficiency	87%
Turbine B efficiency	87%
Main compressor efficiency	68%
Recompressor efficiency	68%
Alternator rated power	123 [kW]
LT recuperator	610 [kW]
HT recuperator	2232 [kW]

Table 2. State points and operational parameters measured at steady operation. Calculated columns include (a) model benchmarking, (b) cycle operation at design conditions, (c) insulation improvements to reduce heat loss, and (d) compressor design improvements to reach 80% efficiency.

Location	Measured (Figure 4, 7600 s)	Calculated (a) benchmark	Calculated (b) design conditions	Calculated (c) insulated	Calculated (d) efficient compressor
Temperature [°C]					
1a-A	32.3	31.9	32.4	32.4	32.4
1b-A ^d	31.8	31.9	32.4	32.4	32.4
2a-A	46.8	47.6	52.6	52.5	51.6
2b-A	45.7	47.6	52.6	52.5	51.6
3a-A	55.8	52.7	119.6	120.8	116.3
3a-B	69.5	71	115.5	116.4	110.2
3b	58.7	55.8	117.7	118.8	113.5
4a	331.7	316.7	422.7	435	432.9
4b	316.1	301.1	407.1	435	432.9
5a	400.4	400.4	538	538	538
5b-A	390	394.1	531.7	538	538
5b-B	390.6	394.6	532.2	538	538
6a-A	368.9	368.5	473.7	484.6	484.5
6a-B	368.9	367.7	473.8	484.1	484
6b	366.3	365.5	471.1	484.3	484.2
7a	67	61	135.8	137.8	132.2
7b	67	61	135.8	137.8	132.2
8a	45.9	47.7	64.3	64.2	63
8b-A	44.6	47.7	64.3	64.2	63
8b-B	45.1	46.9	63.5	64.2	63
Pressure [kPa]					
1a-A	7530.7	7485	7688	7688	7688
1b-A ^d	7485.1	7485	7688	7688	7688
2a-A	10296.9	10218.6	14091.3	14078.3	14099.7
2b-A	10156.8	10218.6	14091.3	14078.3	14099.7
3a-A	10067.4	10136.8	13978.5	13965.6	13986.9
3a-B	10020.4	10126.1	13963.1	13967.1	13980.2
3b	10017.3	10126.1	13963.1	13967.1	13980.2
4a	10117.1	10045.1	13851.4	13855.4	13868.3
4b	10045.3	10045.1	13851.4	13855.4	13868.3
5a ^d	9898.5	9797.8	13472.3	13477.4	13489.7
5b-A	9893.7	9797.8	13472.3	13477.4	13489.7
5b-B	9852.6	9797.8	13472.3	13477.4	13489.7

6a-A	7938.4	7933.7	8416.4	8413.6	8415.3
6a-B	7962.5	7933.7	8416.4	8413.6	8415.3
6b	7983.2	7933.7	8416.4	8413.6	8415.3
7a	7844.5	7808.8	8283.8	8281.1	8282.7
7b	7922.2	7808.8	8283.8	8281.1	8282.7
8a	7783	7685.8	8153.4	8150.7	8152.3
8b-A	7553.1	7685.8	8153.4	8150.7	8152.3
8b-B	7573.5	7685.8	8153.4	8150.7	8152.3
Mass flow [kg/s]					
2a-A	2.506	2.767	3.098	3.134	3.071
3a-B	0.977	0.888	2.534	2.481	2.554
4b	3.483	3.655	5.632	5.615	5.625
5b-A	1.741	1.803	2.626	2.619	2.623
5b-B	1.741	1.854	3.007	2.996	3.001
6a-A	1.712	1.774	2.597	2.589	2.593
6a-B	1.716	1.829	2.982	2.972	2.977
8b-A	2.535	2.797	3.127	3.164	3.101
8b-B	1.002	0.912	2.559	2.506	2.578
TAC A					
RPM ^d	36550	36550	75000	75000	75000
Windage ^a [kW]	N/A ^e	4.9	25.8	25.8	25.8
Turbine heat loss ^b [kW]	8.1	10.5	16.0	0.0	0.0
Turbine power (gross) [kW]	36.4	36.9	151.8	153.7	154.2
Turbine efficiency [%]	79.1	80.8	84.6	84.6	84.6
Compressor pressure ratio	1.376	1.365	1.833	1.831	1.834
Compressor power [kW]	39.2	25.5	47.5	47.9	36.4
Compressor efficiency [%]	36.3	67.8	67.1	67.3	87.1
TAC A net power [kW]	10.9	6.5	78.5	80.0	92.0
TAC B					
RPM ^d	46025	46025	75000	75000	75000
Windage ^a [kW]	N/A ^e	10.1	25.8	25.8	25.8
Turbine heat loss ^b [kW]	8.1	10.8	18.3	0.0	0.0
Turbine power (gross) [kW]	37.8	40.8	176.0	178.0	178.6
Turbine efficiency [%]	84.6	86.5	85.4	85.3	85.3
Compressor pressure ratio	1.323	1.318	1.713	1.714	1.715
Compressor power [kW]	16.7	14.9	93.2	91.6	72.2
Compressor efficiency [%]	61.9	63.4	68.3	68.7	88.6
TAC B net power [kW]	5.7	15.8	57.0	60.6	80.6
System Parameters					
Heater input [kW]	341.7	422.6	891.0	700.9	716.5
Pre-cooler [kW]	239.2	240.2	476.4	481.3	464.8

LT recuperator UA [kW/K]	67.6	67.6	70.0	70.0	70.0
HT recuperator UA [kW/K]	55.2	55.2	40.0	40.0	40.0
Mass loading [kg]	101.2	101.8	104.0	103.3	104.2
Heat loss ^c 4a to 4b [kW]	62.3	65.8	104.9	0.0	0.0
Heat loss ^c 5a to 5b [kW]	40.9	25.8	41.5	0.0	0.0
Heat loss ^c 6a to 6b [kW]	10.5	10.8	17.2	0.1	0.1
Net electricity [kW]	16.6	22.3	135.5	140.6	172.6
Cycle efficiency [%]	4.9	5.3	15.2	20.0	24.1

^aObtained using data presented in Figure 6

^bObtained following section 3

^cObtained following the approach in section 2.3.1

^dInput parameter for model

^eNot directly measured

Figure 1. Layout of split-flow recompression Brayton cycle components.

(a)

(b)

Figure 2. (a) Typical current system operating conditions at Sandia National Laboratories and (b) projected system operation by Dostal *et al.* [30].

Figure 3. Schematic of the internals for the Sandia sCO₂ Turbo-alternator-compressor.

Figure 4. System response to 50% reduction in power setting lasting approximately one minute and two and a half minutes. (a) Low pressure response, (b) high pressure response, (c) transient source boundary temperatures, and (d) system net power generation response (negative power indicates power production from the system).

Figure 5. System response to 100% reduction in power setting lasting approximately one minute and two minutes. (a) Low pressure response, (b) high pressure response, (c) transient source boundary temperatures and, (d) system net power generation response (negative power indicates power production from the system).

Figure 6. Breakdown of alternator windage losses for CO₂ at 27 °C and 1.4 MPa. Each data set was determined by using turbulent correlations and then summed to illustrate the relative contribution of the components.

(a)

(b)

Figure 7. (a) First law efficiency as a function of heat exchanger (HX) effectiveness and (b) exergetic efficiency as a function of heat exchanger ΔT for direct (e.g. sCO_2 in receiver) and indirect (e.g. salt or secondary media in receiver) approaches for several storage capacities at 700 °C, assuming a 98% storage efficiency and 8 hours of daylight operation.